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International application number: PCT/GB05/000120

International filing date: 14 January 2005 (14.01.2005)

Document type: Certified copy of priority document

Document details: Country/Office: GB
Number: 0400794.4
Filing date: 14 January 2004 (14.01.2004)

Date of receipt at the International Bureau: 17 February 2005 (17.02.2005)

Remark: Priority document submitted or transmitted to the International Bureau in compliance with Rule 17.1(a) or (b)

PCT/GB 2005 / 0 0 0 1 2 0

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2. Patent application number *(The Patent Office will fill this part in)* — 0400794.4 — 14 JAN 2004

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LOTUS CARS LIMITED
HETHEL
NORWICH
NORFOLK NR14 8EZ
UNITED KINGDOM

Patents ADP number *(if you know it)* — 5739743002

If the applicant is a corporate body, give the country/state of its incorporation — UK

A/C 13/2/04

4. Title of the invention — A TWO-STROKE COMPRESSION IGNITION ENGINE

5. Name of your agent *(if you have one)* — BOULT WADE TENNANT

"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*

VERULAM GARDENS
70 GRAY'S INN ROAD
LONDON
WC1X 8BT

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MR A.W. PLUCKROSE

020-7430-7500

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A TWO-STROKE COMPRESSION IGNITION ENGINE.

The present invention relates to a two-stroke compression ignition (commonly called "diesel") engine.

For use in automobiles, high speed direct injection (HSDI) compression ignition engines have been developed and have proved very successful. However, the diesel combustion process is inherently speed-limited. The four-stroke HSDI diesel engines currently available currently have a maximum speed of revolution of 5000 rpm, which limits their specific power output to approximately 75 bhp per litre.

The present invention provides: a two-stroke compression ignition internal combustion engine comprising:

a cylinder with a cylinder axis, the cylinder formed as a bore in a cylinder block and having a cylinder head;

a piston reciprocating axially along the cylinder and defining with the cylinder a variable volume combustion chamber;

injection means for injecting fuel into the variable volume combustion chamber;

a transfer port in the cylinder bore which is covered and uncovered by the piston in a cyclical manner during reciprocation of the piston in the cylinder;

exhaust valve means in the cylinder head;

actuator means for controlling opening and closing of the cylinder head exhaust valve means;

an electronic controller controlling operation of

the actuator means; and

compressor means for compressing air for delivery to the combustion chamber as scavenge and charge air; wherein in a two-stroke cycle of the engine:

at commencement of an upstroke of the piston the transfer port is open to the combustion chamber and air compressed by the compressor means is delivered via the transfer port to the combustion chamber, in the upstroke the piston covers the transfer port and then the delivered air is compressed in the combustion chamber, during the upstroke fuel is injected into the combustion chamber by the injection means and the resulting fuel/air mixture in the combustion chamber is ignited by compression injection; and

at commencement of a downstroke of the piston the combustion of the fuel/air mixture causes gases in the combustion chamber to expand and force the piston downwardly, the controller operates the actuator means to open the cylinder head exhaust valve means to allow the combusted gases to be exhausted from the combustion chamber and in the last part of the downstroke the transfer port is uncovered by the piston and air compressed by the compressor means is delivered via the transfer port to the combustion chamber to scavenge the combusted gases out of the combustion chamber via the open cylinder head exhaust valve means.

The applicant has realised that a two-stroke operating regime is ideally suited to the use of compression ignition in a reciprocating piston internal combustion engine.

Operating a compression ignition engine with a two-stroke cycle offers a specific power output significantly higher than that of a four-stroke compression ignition engine (the theoretical output of the two-stroke engine could approach double that of the equivalent four-stroke engine). Also, the compression ignition engine of the present invention solves the hydrocarbon emission problems typical with spark ignition two-stroke engines, in which fuel often passes directly from the transfer ports to the exhaust ports, and is exhausted without ignition in a combustion chamber. In the present invention the cylinder head exhaust valve means is operated in timed relationship with the injection means so that injected fuel is not swept out of the combustion chamber via an open exhaust port. Also, preferably the injector is mounted in the cylinder head and so sprays away from the cylinder head exhaust ports.

The engine of the present invention uses uniflow scavenging rather than simple loop-scavenging or cross-scavenging. This is a more efficient process giving a better delivery ratio (i.e. volumetric efficiency) and in theory a higher maximum power output. The scavenge air is cooler than the exhaust gases and so introduction of the air into the combustion chamber via one or more transfer ports in the cylinder bore (rather than via cylinder head ports) avoids thermal distortion problems in the cylinder bore, the scavenge air having the effect of cooling the cylinder bore.

Preferred embodiments of the present invention will now be described with reference to the accompanying drawings in which:

Fig. 1 is a cross section through a two-stroke compression ignition reciprocating piston internal combustion engine according to the present invention;

Fig. 2 is a schematic illustration of a three-cylinder two-stoke compression ignition internal combustion engine according to the present invention arranged with a low pressure turbocharger and a high pressure turbocharger;

Fig. 3 is a schematic illustration of a three-cylinder two-stroke compression ignition internal combustion engine according to the present invention with an electrically driven low pressure compressor and a turbocharger;

Fig. 4 is a schematic illustration of a three-cylinder two-stroke compression ignition internal combustion engine according to the present invention having a low pressure turbocharger and a high pressure turbocharger;

Fig.5 shows schematically how the Fig. 4 three-cylinder two-stroke compression ignition engine can be operated in a manner which assists starting of the engine;

Figure 6 shows schematically how the Figure 4 three-cylinder two-stroke compression ignition engine could be modified to provide for storage of compressed gas; and

Figure 7 is a cross-section of the Figure 1 engine showing a variable angle vane mechanism which could be incorporated in the engine.

There will now be described a number of variants of two-stroke compression ignition reciprocating piston internal combustion engines according to the present invention. All of the designs have common features. Each has a power target of 130 bhp and 260 Nm per litre, with a

mass of less than 165kg (not including the engine management system, catalyst or harness). This would be achieved with a 1.5 litre capacity, using three cylinders with a balancer shaft. The bore of each cylinder would be 84mm with a stroke of 90mm (498.8cc per cylinder). The engines should be able to produce 195 bhp at 4000 rpm (14.6 bar BMEP) and 390 Nm at 2000 rpm (16.4 bar BMEP). They should have a maximum engine speed of 4750 rpm (14.3 m/s mean piston speed). Diesel fuel will preferably be injected into the cylinders by direct injection using a common rail system. The fuel will be ignited by a homogeneous charge compression ignition ('HCCI') process in some operating conditions of the engine and by usual compression ignition (also known as "diesel" ignition) in other operation conditions. The engines will preferably be formed with a monoblock construction with aluminium main castings, thus removing the need for a cylinder head gasket.

Fig. 1 shows in schematic form one cylinder of a three-cylinder two-stroke compression ignition engine according to the invention. The figure shows a piston 10 reciprocable in a cylinder defined in a cylinder bore 11, the cylinder and the piston together defining a variable volume combustion chamber 12. A ring of transfer ports is provided at the base of the cylinder bore. The transfer ports need not be equi-spaced from each other, but could be clustered in e.g. two groups. Two transfer ports 13, 14 are illustrated in Figure 1. Four cylinder head exhaust valves are also provided. Two of these valves 15, 16 are shown in the figure. Each of the exhaust valves is connected to a hydraulic actuator.

Separate actuators can be provided for each exhaust valve or for one actuator can operate a pair of exhaust valves. An actuator 17 is shown operating the exhaust valve 15 and an actuator 18 is shown operating the exhaust valve 16. Each of the exhaust valves 15, 16 are poppet valves provided in the cylinder head of the cylinder defined in the block 11.

The actuator 17 is connected via an electrically controlled servo-valve 21 to a source of pressurised fluid 19 and an exhaust for pressurised fluid 20 (typically a sump from which the source 19 draws fluid). In a similar way the actuator 18 is connected via an electrically controlled servo-valve 22 to the source 19 and exhaust 20. Each of the electrically controlled servo-valves 21, 22 can control opening and closing of the exhaust valves 15, 16 by controlling flow of hydraulic fluid to and from the chambers of the actuators 17, 18, in a known manner. Preferably, the electrically controlled servo-valves 21, 22 will also be able to control the rate of flow of fluid to and from the actuators 17, 18 in order to control the speed of opening and closing of the exhaust valves 15, 16.

A common rail diesel injector 23 is placed centrally in the cylinder head, for the best combustion chamber geometry (as is known with existing four-stroke HSDI engines). The injector 23 need not be part of a common rail system, but is preferably a high pressure multiple injection event injector.

The servo-valves 21,22 and the injector 23 are controlled by an electronic controller 30 of an engine management system.

Transfer ports (e.g. 13, 14) are angled in order to provide an angled delivery of charge air into the cylinder. In this way a very regimented, repeatable air motion can be set up in the cylinder, typically a 'swirl' centred on the cylinder axis. This is vital for direct injection diesel combustion. Most four-stroke diesel engines suffer from compromises in the cylinder head architecture to achieve regimented repeatable air motion. This is not necessary in the engine of the present invention, because the correct transfer port slot angles can provide the necessary air flow, as will be described later. In a variant of the engine of the present invention shown in cross-section in Figure 7, the transfer ports are provided by two apertures, 80, 81, having in each a plurality of vanes 82, 83, 84 and 85, 86, 87. The vanes are mounted on a ring 88 which has a toothed section 89. The toothed section 89 is engaged by a gear 90 which is driven by a stepper motor (not shown) under control of the controller 30. The ring 88 can be rotated to vary in orientation the vanes 82 to 87 and thus control swirl in the combustion chamber 12. The pressurised air is supplied via passages 91, 92 in the direction of the arrows shown, to aid swirl motion in the chamber 12.

To facilitate scavenging of the engine a forced induction system will be provided to pressurise charge air admitted to the cylinders. Various possibilities are possible and these are described later with reference to Figs. 2 to 6.

A two-stroke cycle of the piston and cylinder illustrated in Fig. 1 will now be described, starting with the piston 10 at its bottom dead centre (BDC) position. In this position the transfer ports (e.g. 13, 14) will be

uncovered by the piston 10 and pressurised charge air will be forced into the cylinder in order to scavenge from the cylinder previously combusted gases. The combusted gases will be scavenged to exhaust through cylinder head exhaust ports opened by the cylinder head exhaust valves (e.g. the poppet valves 15, 16). As the piston 10 moves upwardly it will close the transfer ports, e.g. 13, 14. During the upstroke of the piston 10 the actuators 17, 18 will act to close the exhaust valves 15, 16. The air trapped in the variable volume combustion chamber 12 is then compressed by the continued upward motion of the piston 10. The trapped air will have a significant swirl component to its motion, this being occasioned by the angled nature of the transfer ports, e.g. 13, 14. Into the swirling trapped air diesel fuel is injected by the injector 23. The swirling air will ensure a good mixing of the diesel fuel with the charge air in the combustion chamber 12. The mixture of fuel and air can be combusted either through homogeneous charge compression ignition (HCCI) at part load conditions of the engine or through conventional combustion ignition (also known as "diesel" ignition) combustion in other operating conditions of the engine. The injector 23 will be a pulsed injector injecting fuel in pulses into the chamber 12. For HCCI the pulses will start early enough in the upstroke for the fuel and air to form a homogeneous mixture prior to ignition.

In the downstroke of the piston the combusted fuel and air mixture will expand due to the combustion and force the piston 10 to move downwardly. In the later part of the downstroke of the piston 10 the actuators 17, 18 will be controlled to open the cylinder head exhaust valves, e.g.

15, 16 to allow the combusted gases to escape from the combustion chamber 12 to exhaust. The downwardly moving piston 10 will then uncover the transfer ports, e.g. 13, 14 and pressurised charge air will be introduced into the combustion chamber 12 to scavenge from the chamber 12 the previously combusted gases to exhaust.

The scavenging process in the engine is a uniflow scavenging process. Uniflow scavenging gives a more efficient scavenging of a cylinder than simple loop-scavenging or cross-scavenging. It is advantageous that the fresh scavenge air is introduced through ports in a cylinder bore rather than through ports in a cylinder head. It is advantageous because the scavenge air is cooler than the exhaust gases and therefore thermal distortion problems in the cylinder bore are minimised.

To assist HCCI combustion the controller 30 can close the exhaust valves e.g. 15, 16 early in an upstroke of the piston in order to trap combusted gases in the combustion chamber 12. The trapped combusted gases advantageously inhibit the ignition of the fuel/air mixture to delay combustion. The amount of combusted gases trapped in the chamber 12 can be varied by varying closing of the valves 15, 16 and thus the commencement of ignition controlled to a limited degree. However, HCCI is only possible on low and part loads and at high loads it will be necessary to run the engine using conventional compression ignition.

As mentioned above, the charge air will need to be

pressurised by a forced induction system. The total backpressure at the exhaust valves, e.g. 15, 16, must be lower than the pressure of the pressurised charge air at the point that the transfer ports, e.g. 13, 14, are opened, or else the engine will not scavenge. It is envisaged that the total exhaust port pressure will be around 2.1 bar absolute and therefore the compressed charged air must be pressurised to 2.5 bar absolute. This calculation is based upon a requirement for boost of 1.1 - 1.3 bar gauge in order that the BMEP approaches the levels achievable in current four-stroke HSDI engines. These requirements raise issues for a forced induction system in a two-stroke engine which are not present in a four-stroke engine. In particular, it is unlikely for reasons of fuel economy that a diesel two-stroke will want to rely solely on a full-flow-operation supercharger. Furthermore, for a two-stroke engine there are problems faced on starting and at low engine speeds, when it is not easy to achieve the necessary pressure in the scavenge air by turbo-charging. Different solutions to these problems are now discussed with reference to the attached drawings.

Fig. 2 shows schematically the three cylinder two-stroke compression ignition internal combustion engine 100 according to the present invention, with a forced induction system comprising a low pressure stage having a turbo-charger 101 and a high pressure stage having a super-charger 102. In the figure three cylinders 103, 104 and 105 are shown, each of which will be a cylinder as shown in Fig. 1. Each cylinder has a pair of exhaust valves "a" which control

flow of exhaust gas via a passage 109 to a turbine of the low pressure turbocharger 101. Each cylinder also has a pair of exhaust valves "b" which control flow of exhaust gas to a bypass passage 103. The bypass passage 103 allows exhaust gas to flow straight to atmosphere bypassing the low pressure turbocharger 101.

The Fig. 2 engine works with charge air being drawn in via an air filter 104 into the compressor part of the low pressure turbocharger 101. The pressurised air then flows out via a passage 105 to a bypass valve 106 or to the compressor part of a high pressure supercharger 102. Then the charge air pressurised in the high pressure supercharger 102 flows out through the passage 107. The bypass valve 106 could be controlled by the engine management system to control the amount of pressurised charge air flowing into the compressor of the supercharger 102. Alternatively, it could be a simple mechanical pre-loaded valve which would open at a defined pressure to limit the pressure of the scavenge air flowing as an input to the compressor of the supercharger 102. The bypass scavenge air and the pressurised air exiting the supercharger 102 are mixed before they flow through an intercooler 108 and then on to the cylinders 103, 104, 105.

The engine management system controls the opening of the exhaust valves "a" and "b" in each cylinder to control the amount of pressurised exhaust gas flowing to the turbine of the low pressure turbo charger 101. A portion of the exhaust gas is allowed to flow to the turbine of the turbo charger 101 and a portion is allowed to flow via the bypass passage 103 directly to atmosphere.

It is envisaged that the supercharger 102 would typically be a Roots blower type supercharger. It could be a clutched supercharger so that it is operated only in certain engine operating conditions, under control of the controller 30.

A second variant of engine is shown in Fig. 3. Again an engine 200 is shown with three cylinders each of the type shown in Fig. 1. Again, each cylinder has four cylinder head exhaust valves operated in pairs. Each cylinder has a pair of exhaust valves "a" connected to a first exhaust duct 201 and each cylinder has a pair of exhaust valves "b" connected to a second exhaust duct 202 separate from the first exhaust duct 201.

In the Fig. 3 engine fresh air is drawn in via a filter 204 to be compressed by an electrically powered compressor 205. The electrically powered compressor 205 is controlled by controller 30 to operate at low speeds of the engine and/or during starting, but does not operate otherwise. In other conditions a bypass valve 206 is opened to allow charge air to bypass the low pressure electrically driven compressor 205.

Air exiting the low pressure compressor 205 or passing through the bypass valve 206 then flows on to a high pressure turbocharger 207 to be compressed in the turbocharger and then output via a duct 208 to an intercooler 209 and then on to the transfer ports of the cylinders of the engine 200.

Combusted gases can be exhausted from the cylinders 210, 211, 212 either via the exhaust valves "a" or by the exhaust valves "b". These valves are controlled by actuators controlled by an engine management system. The engine management system will control operation of the valves "a" and "b" to control what proportion of the exhaust gases flow through the exhaust duct 201 and what proportion flow through the exhaust duct 202. The exhaust gases flowing through the exhaust duct 201 flow to the turbine of the high pressure turbo charger 207, whilst the exhaust gases flowing through the exhaust duct 202 bypass the turbocharger 207 and flow directly to atmosphere.

In Fig. 4 a third variant of forced induction system is shown, having a pair of turbochargers comprising a low pressure turbocharger 301 and a high pressure turbocharger 302. The engine 300 shown in Fig. 4 is again of the same type of the previously described engines, having three cylinders 303, 304 and 305 each having a first pair of exhaust cylinder head exhaust valves "a" connected to a first exhaust duct 306 and each having a second pair of cylinder head exhaust valves "b" connected to a second exhaust duct 307 independent and separate from the first exhaust duct 306.

Air is drawn into the engine 300 via an air filter 308 and into the compressor part of the low pressure turbo charger 301. The pressurised charge air is output to a duct 309 to pass either to a bypass valve 310 or to pass into the compressor part of a high pressure turbocharger 302.

The bypass valve 310 can be controlled electrically by the engine management system to control how much of the pressurised charge air flows through the high pressure turbocharger 302 and how much bypasses the turbocharger 302. Alternatively, the bypass valve 310 can be a simple mechanical valve which limits the pressure of the charge air flowing into the compressor of turbocharger 302. The charge air compressed by the turbocharger 302 combines with any bypass air and then passes into an intercooler 311 and then onto the engine 300 to be used as scavenge air and charge air. Combusted gases resulting from combustion in the cylinders 303, 304, 305 are output to the exhaust ducts 306, 307 under the control of the cylinder head exhaust valves "a" and "b" in each cylinder, which are in turn operated by the hydraulic actuators described in Fig. 1, with the engine management system controlling operation of the actuators to control the proportion of exhaust gases flowing in ducts 306 and the proportion flowing in ducts 307.

The exhaust gases that are flowing in duct 306 pass to the turbine section of the high pressure compressor 302. The exhaust gases in duct 307 flow to the turbine part of the low pressure turbocharger 301. The exhaust gases exiting the turbine part of the high pressure turbocharger are fed into the exhaust gases flowing in exhaust duct 307 to then flow to the turbine part of the turbo charger 301. Exhaust gases flowing out of the turbine part of turbo charger 301 flow through the exhaust duct 312 to atmosphere. All of the exhaust gases flow through the turbine of turbocharger 301

Fig. 5 shows how the engine of Fig. 4 can be beneficially modified to assist starting of the engine. Since the engine illustrated has most components in common with the engine of Fig. 4 the same reference numerals have been used. The additional feature of the engine is the starting valve 320. This will be controlled by the engine management system. During engine starting the starting valve 320 will be closed. Also the controller will vary the operation of the exhaust valves. By closing the valve 320 and varying operation of the exhaust valves the controller can arrange the engine to operate such that gas is compressed in each of the combustion chambers and then expelled via the exhaust valves "a". The expelled gas powers the high pressure turbocharger 302 and starts it spinning. The gas exhausted from the turbine of the turbocharger 302 is then fed back into the combustion chambers via the exhaust valves "b". The gas that is fed back in is then pressurised again, let out by the exhaust valves "a" and the cycle is repeated. This enables the engine to work as a pneumatic pump to start the high pressure turbo charger 302 spinning rapidly prior to injection of fuel into the combustion chambers and starting of the engine. This is very beneficial, particularly since the recirculated air will be hotter than fresh charge air. Providing this facility removes the need for a supercharger or an electrically driven compressor, which would be typically chosen to assist starting of a compression ignition engine not having the fast start mode of operation illustrated in Fig. 5.

Whilst the Fig. 5 arrangement for fast start operation systems is shown applied to the engine illustrated in Fig. 4 it is possible that the engines of Fig. 2 or Fig. 3 could be arranged to provide fast start modes with the gases leaving the turbo chargers 101 and 207 recycled via the exhaust valves "b" into the combustion chambers for further compression.

Figure 6 shows a further variant of the Figure 4 engine. In this variant each cylinder has only one exhaust valve "a", but an additional type of exhaust valve "c". The exhaust valves "a" and "b" will be operated as described before, save during engine braking and engine starting when the valve "c" may be used. The additional exhaust valves "c" are connected via passages 601,602,603 to a storage tank 604 for storing compressed gases. The valves "c" are controlled during engine braking to allow compressed gases to flow from the cylinder to the storage tank 604. The valves "c" can then be opened when needed (e.g. on starting of the engine) to supply previously stored compressed gases to the cylinder, e.g. to expand in the cylinder and drive the pistons to reciprocate.

The valves "c" are operated to allow flow of compressed gas to the storage tank 604 only when the tank is not already pressurised to its limit. The valves "c" allow flow of gas from the storage tank 604 to the cylinders only when the pressure in the storage tank 604 is sufficient.

The use of actuators to open and close the poppet valves in the cylinders is very important. However, the achievement of poppet valve operation at two-stroke operation rates is problematic. Use of cams and tappets severely speed limits a two-stroke engine. By using hydraulic actuators the two-stroke engines proposed above will be able to handle higher operation speeds. They will not be limited by cycle speed in the same way that a conventional cam-base system is. Additionally, the use of actuators for the cylinder head exhaust valves can allow optimised expansion/blow down (for best thermal efficiency) and also can optimise HCCI operation (for minimum NOx output from the engine). Furthermore, if required, the cylinder head exhaust valves can be operated to provide a wastegate function at full load, in the event that the charge air supplied is excessively pressurised. The use of actuators therefore yields specific advantages in engine speed, HCCI operation of the engine, thermal efficiency, simplified start procedure and turbo charger boost control. Above hydraulic actuators are described but other actuators such as electro-magnetic actuators could be used.

The use of poppet valves offers a solution which is mechanically sound and allows high cylinder gas loads. However, it is possible that other valves such as sleeve valves could be used instead of poppet valves.

In the embodiment of the engines shown in Figs. 4 and 5 it is possible that the lower pressure turbo charger could be replaced with an electrically-assisted turbocharger,

which is assisted by electrical power at low engine speeds or on starting, but is otherwise powered by the exhaust gases from the engine. An electrically-assisted turbocharger could be used to output electrical power at high engine speeds.

The turbochargers used could be fixed geometry or variable geometry turbochargers.

CLAIMS

1. A two-stroke compression ignition internal combustion engine comprising:

a cylinder with a cylinder axis, the cylinder formed as a bore in a cylinder block and having a cylinder head;

a piston reciprocating axially along the cylinder and defining with the cylinder a variable volume combustion chamber;

injection means for injecting fuel into the variable volume combustion chamber;

a transfer port in the cylinder bore which is covered and uncovered by the piston in a cyclical manner during reciprocation of the piston in the cylinder;

exhaust valve means in the cylinder head;

actuator means for controlling opening and closing of the cylinder head exhaust valve means;

an electronic controller controlling operation of the actuator means; and

compressor means for compressing air for delivery to the combustion chamber as scavenge and charge air; wherein in a two-stroke cycle of the engine:

at commencement of an upstroke of the piston the transfer port is open to the combustion chamber and air compressed by the compressor means is delivered via the transfer port to the combustion chamber during the upstroke, in the upstroke the piston covers the transfer port and then the delivered air is compressed in the combustion chamber, during the upstroke fuel is injected into the combustion

chamber by the injection means during the upstroke of the piston and the resulting fuel/air mixture in the combustion chamber is ignited by compression injection; and

at commencement a downstroke of the piston the combustion of the fuel/air mixture causes gases in the combustion chamber to expand and force the piston downwardly, then the controller operates the actuator means to open the cylinder head exhaust valve means to allow the combusted gases to be exhausted from the combustion chamber and in the last part of the downstroke the transfer port is uncovered by the piston and air compressed by the compressor means is delivered via the transfer port to the combustion chamber to scavenge the combusted gases out of the combustion chamber via the open cylinder head exhaust valve means.

2. A two-stroke compression ignition internal combustion engine as claimed in claim 1, wherein the transfer port is angled to provide the delivered air with a swirl motion.

3. A two-stroke compression ignition internal combustion engine as claimed in claim 2, comprising a plurality of transfer ports arranged in a ring around the cylinder bore, each covered and uncovered by the piston in a cyclical manner during reciprocation of the piston in the cylinder bore such that compressed air delivered therethrough has a swirl motion in the combustion chamber.

4. A two-stroke compression ignition internal combustion engine as claimed in any one of claims 1 to 3 wherein the or

each transfer port has at least one vane pivotally mounted in the engine and control means for pivoting the vane to vary the degree of swirl imparted to air passing through the or each transfer port.

5. A two-stroke compression ignition internal combustion engine as claimed in any of the preceding claims wherein the injection means comprises a fuel injector positioned centrally in the cylinder head to deliver fuel along the cylinder axis.

6. A two-stroke compression ignition internal combustion engine as claimed in any one of the preceding claims wherein the cylinder head exhaust valve means comprises a plurality of poppet valves.

7. A two-stroke compression ignition internal combustion engine as claimed in any one of the preceding claims wherein the exhaust valve means comprises at least one exhaust valve connected to a first exhaust duct and at least one exhaust valve connected to a second exhaust duct separate and independent from the first exhaust duct.

8. A two-stroke compression ignition internal combustion engine as claimed in Claim 7 wherein: the compressor means comprises a turbocharger and the first exhaust duct is connected to the turbocharger so that the exhaust gases in the first exhaust duct drive the turbocharger to rotate; the second exhaust duct bypasses the turbocharger and relays the exhaust gases flowing therethrough to be exhausted without

passing through the turbocharger; and the controller controls operation of the actuator means to control what proportion of the combusted gases flow through the first exhaust duct and what proportion of the combusted gases flow through the second exhaust duct, the controller thereby controlling operation of the turbocharger.

9. A two-stroke compression ignition internal combustion engine as claimed in Claim 8 wherein: the compressor means comprises additionally a supercharger; the turbocharger is a low pressure turbocharger which compresses intake air to a first pressure; and the supercharger is a high pressure supercharger which compresses the compressed air output by the turbocharger to a second pressure higher that the first pressure.

10. A two-stroke compression ignition internal combustion engine as claimed in Claim 9 wherein the compressor means comprises additionally a bypass passage through which compressed air compressed by the turbocharger can bypass the supercharger; and bypass valve means controlling flow of compressed air through the bypass passage.

11. A two-stroke compression ignition internal combustion engine as claimed in Claim 10 wherein the bypass valve is an electrically-controlled valve controlled by the electronic controller.

12. A two-stroke compression ignition internal combustion engine as claimed in Claim 8 wherein the compressor means comprises additionally an electrically-driven compressor and the turbocharger is a high pressure turbocharger which

received compressed air compressed by the electrically-driven compressor and pressurises the air to a higher level.

13. A two-stroke compression ignition internal combustion engine as claimed in Claim 12 wherein the compressor means compresses additionally a bypass passage through which air can bypass the electrically-driven compressor to flow directly to the turbocharger and a bypass valve controlling flow of air through the bypass passage.

14. A two-stroke compression ignition internal combustion engine as claimed in claim 13 wherein the controller controls operation of the bypass valve and the electrically-driven compressor such that the electrically-driven compressor is operated only on starting the engine and/or at low engine speeds and otherwise intake air bypasses the electrically-driven compressor completely and is compressed only by the turbocharger.

15. A two-stroke compression ignition internal combustion engine as claimed in claim 7 wherein:

the compressor means comprises a low pressure turbocharger which compresses air to a first pressure and a high pressure turbocharger which compresses air compressed by the low pressure turbocharger to a second pressure higher than the first pressure;

the first exhaust duct relays exhaust gas to the high pressure turbocharger to drive the high pressure turbocharger to rotate and the second exhaust duct relays exhaust gas to the lower pressure turbocharger, bypassing the high pressure turbocharger, to drive the low pressure turbocharger to rotate; and

the controller controls operation of the actuator means to control what proportion of combusted gases flowing from the combustion chamber flow through the first exhaust duct and what proportion flow through the second exhaust duct, the controller thereby controlling operation of the high pressure and the low pressure turbochargers.

16. A two-stroke compression ignition internal combustion engine as claimed in claim 15 wherein the expanded exhaust gases leaving the high pressure turbocharger are fed into the second exhaust duct to be relayed to the low pressure turbocharger.

17. A two-stroke compression ignition internal combustion engine as claimed in claim 15 or claim 16 wherein the compressor means comprises additionally a bypass passage through which air can bypass the high pressure turbocharger and a bypass valve controlling flow of air through the bypass passage.

18. A two-stroke compression ignition internal combustion engine as claimed in claim 17 wherein the bypass valve is controlled by the controller.

19. A two-stroke compression ignition internal combustion engine as claimed in any one of claims 8 to 18, wherein the compressor means comprises additionally an intercooler for cooling the compressor intake air prior to delivery of the air into the combustion chamber.

20. A two-stroke compression ignition internal combustion engine as claimed in one of claims 8 to 18, which comprises additionally a starting valve controlled by the controller which can prevent flow of exhaust gases through the second exhaust duct during engine starting and wherein:

exhaust gases leaving the turbocharger supplied by the first exhaust duct are fed into the second exhaust duct upstream of the starting valve; and

the controller during starting of the engine operates to close the starting valve and to open and close the cylinder head exhaust valves so that compressed gases leaving the combustion chamber are relayed via the first exhaust duct to the turbocharger connected thereto to drive the said turbocharger and then are returned to the combustion chamber via the second exhaust duct to be compressed again in the combustion chamber.

21. A two-stroke compression ignition internal combustion engine as claimed in any one of claims 8 to 18 comprising additionally a storage tank, a storage tank passage leading from the cylinder to the storage tank and cylinder head storage tank valve means controlling flow of combusted gases to the storage tank from the cylinder and also flow of stored combusted gases from the storage tank to the cylinder, whereby combusted gases compressed in the cylinder can be relayed to the storage tank for storage therein and for later return to the cylinder for expansion therein.

22. A two-stroke compression ignition internal combustion engine as claimed in any one of the preceding claims wherein

the injector means can inject fuel into the combustion chamber early enough in the upstroke for mixing of the fuel with the air to produce a homogeneous mixture which is then ignited by homogeneous charge compression ignition and wherein the injection means can alternatively inject fuel later in the upstroke for compression ignition in the combustion chamber.

23. A two-stroke compression ignition internal combustion engine as claimed in claim 22 wherein in part load operating conditions of the engine the controller operates to close the cylinder head exhaust valve means during the upstroke of the piston in order to trap combusted gases in the combustion chamber, the trapped combusted gases forming a mixture with the fuel and air and serving to delay ignition of the fuel and air mixture when the engine is operating with homogeneous charge compression ignition.

24. A two-stroke compression ignition internal combustion engine substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

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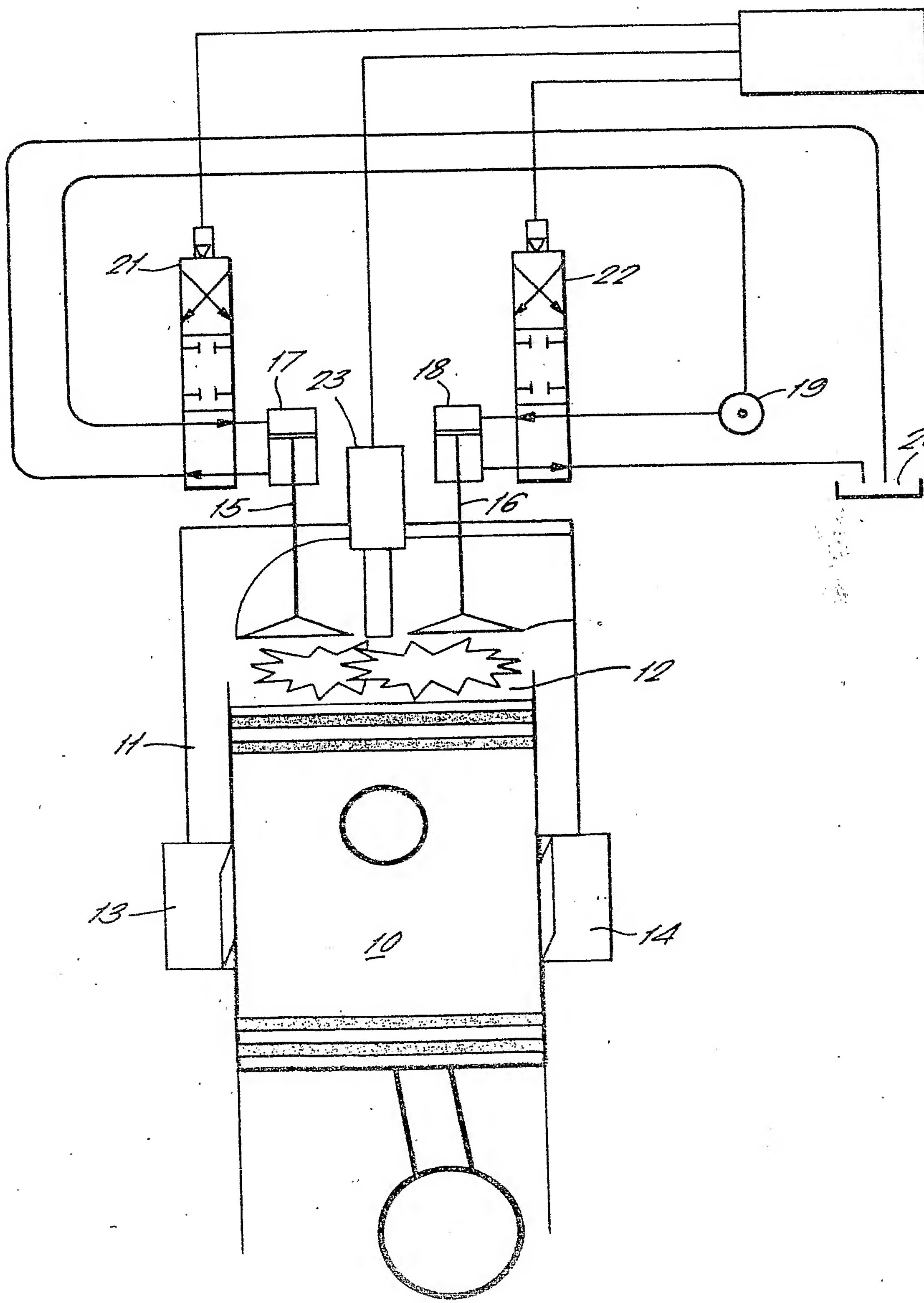
FIG. 1.



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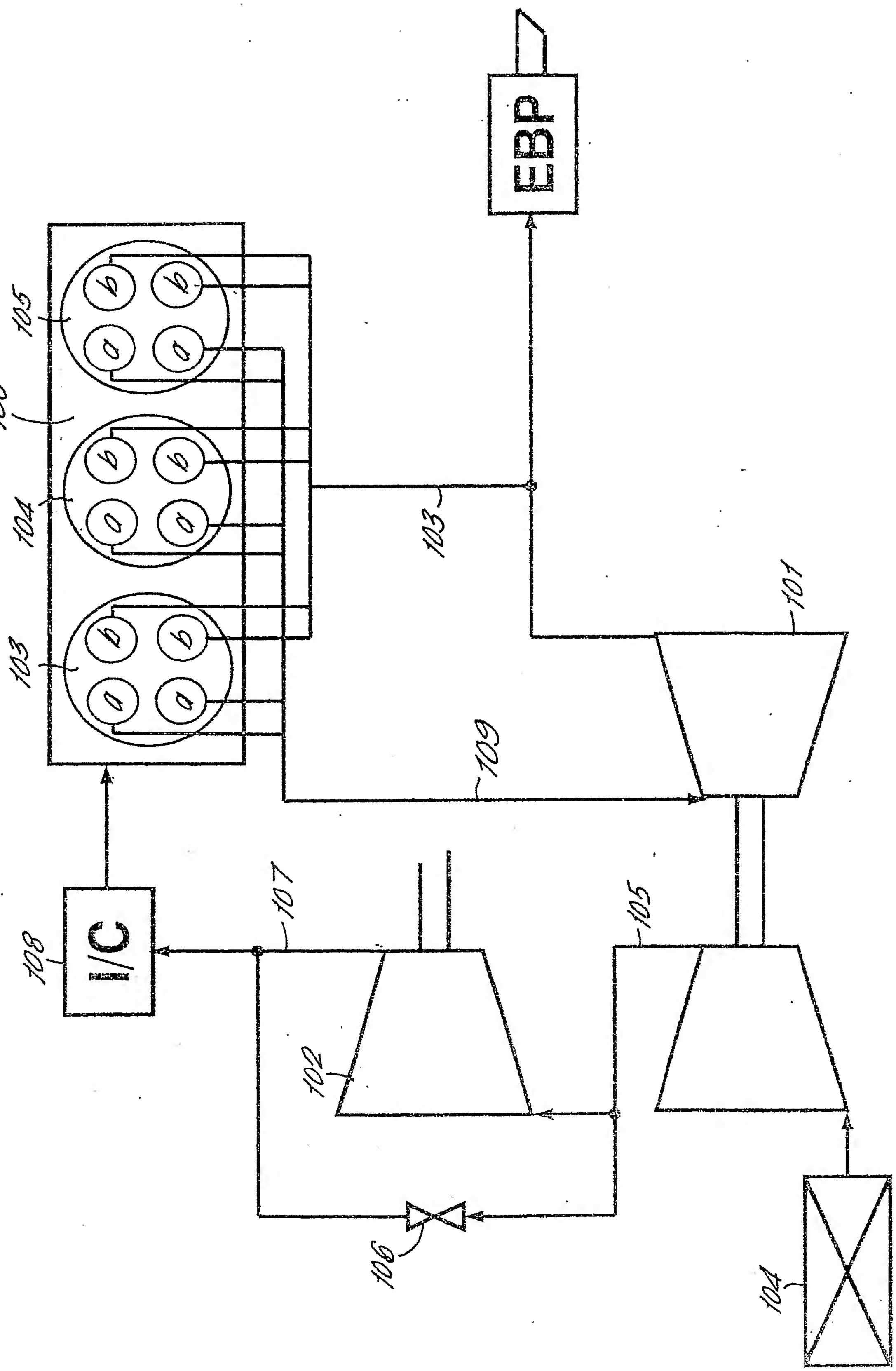

FIG. 2

FIG. 3.

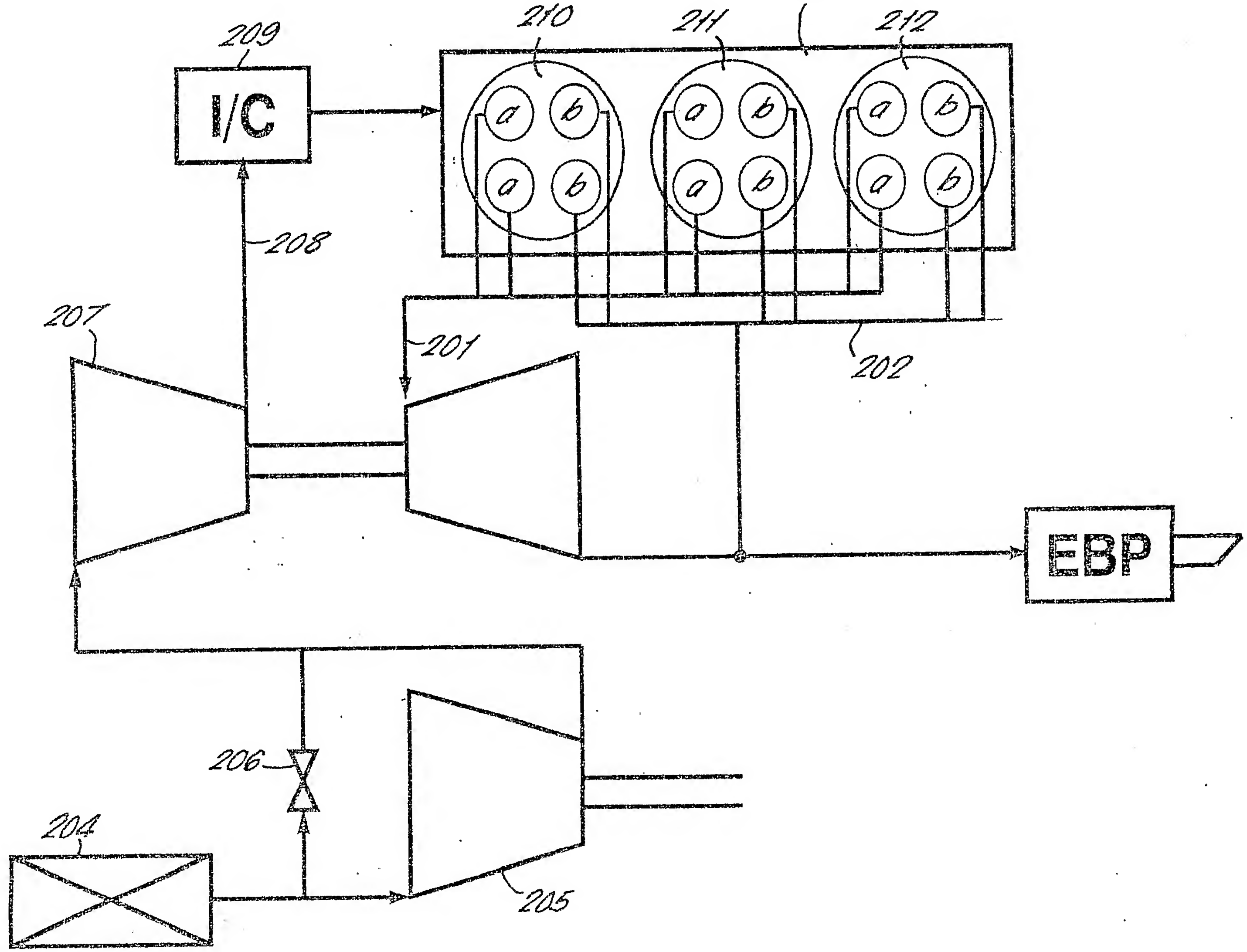

FIG. 4.

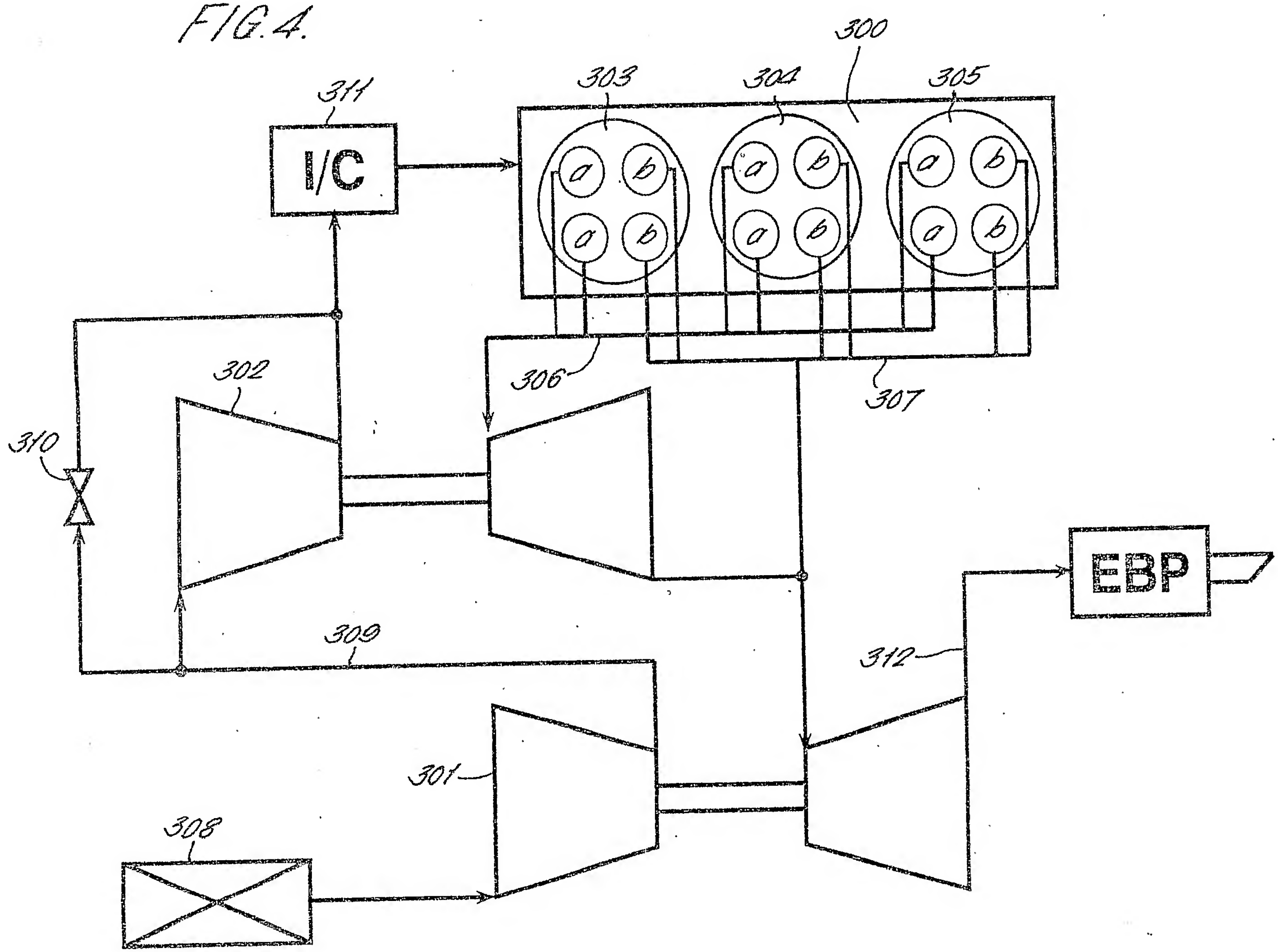

FIG. 5.

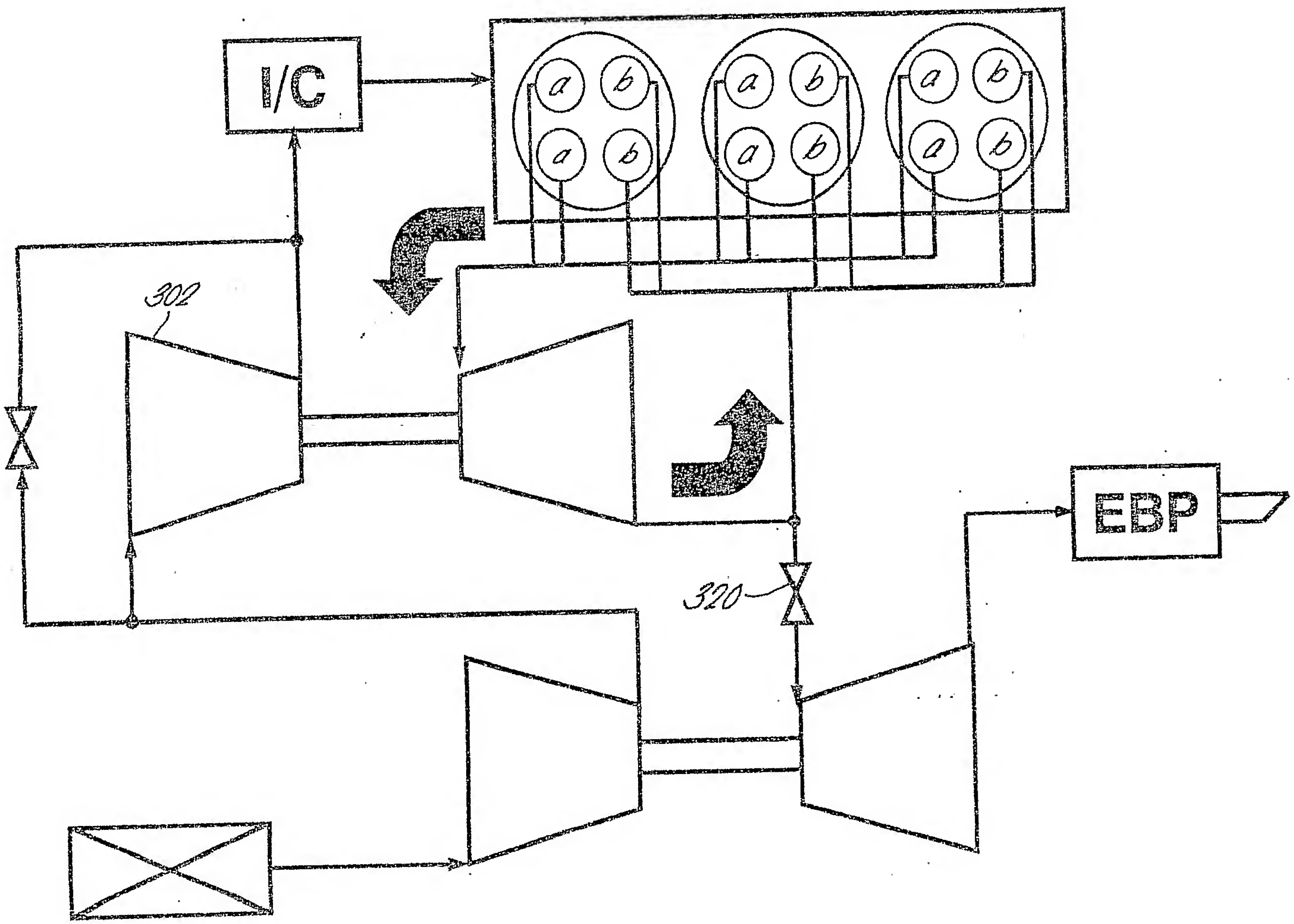

FIG. 6.

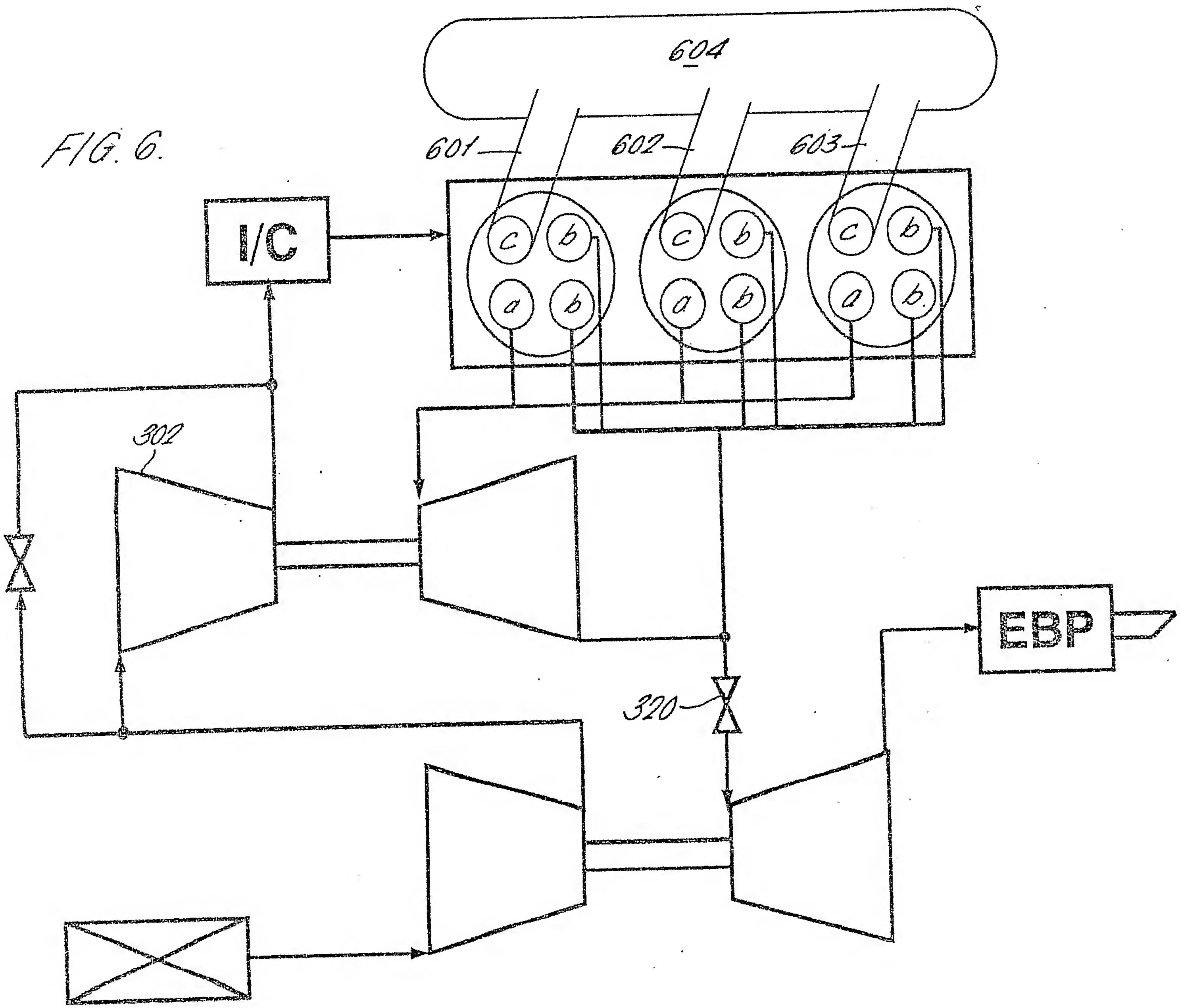

FIG. 7.